Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

2018

Numerical study on the internal characteristics of single screw expanders used in organic Rankine cycle systems

Lili Shen
Beijing University of Technology, China, People's Republic of, 101996sll@163.com

Wei Wang
Beijing University of Technology, China, People's Republic of, wang_wei@bjut.edu.cn

Yu-ting Wu
Key Laboratory of Enhanced Heat Transfer and Energy Conservation, Beijing, China, People's Republic of / Beijing University of Technology, China, People's Republic of, wuyuting1970@126.com

Biao Lei
Beijing University of Technology, China, People's Republic of, leibiao4151@163.com

Ruiping Zhi
Beijing University of Technology, China, People's Republic of, zhiruiping@emails.bjut.edu.cn

See next page for additional authors

Authors
Lili Shen, Wei Wang, Yu-ting Wu, Biao Lei, Ruiping Zhi, and Chongfang Ma

Numerical study on the internal characteristics of single screw expanders used in organic Rankine cycle systems

Li-li SHEN[1,2]*, Wei WANG[1,2], Yu-ting WU[1,2], Biao LEI[1,2], Rui-ping ZHI[1,2], Chong-fang MA[1,2]

[1]Key Laboratory of Enhanced Heat Transfer and Energy Conservation,
Ministry of Education and Key Laboratory of Heat Transfer and Energy Conversion,
Beijing municipality, China

[2]Beijing University of Technology, College of Environmental and Energy Engineering,
Beijing, China
Contact Information (+86 10 67398313, +86 10 67392774, shenlili@emails.bjut.edu.cn)

* Corresponding Author

ABSTRACT

Single screw expanders have been widely studied in middle-low temperature heat recovery power system of organic Rankine cycle, which is significant for the energy-conservation and environment-protection. However, internal irreversible loss including leakage, heat transfer and friction power loss has great influence on the performances of single screw expanders. Although some present researches of single screw expanders have been carried out, few can directly reflect the effect of internal irreversible loss on the performance of expanders. Therefore, it is necessary to research the internal characteristics of the single screw expanders. In this paper, a numerical study of a single screw expander was carried out to analyze its internal irreversible loss. Based on the theory of engineering thermodynamics and hydrodynamics, a thermodynamics working process mathematical model was presented to calculate the flow rate and efficiency of a single screw expander. A separation approach was proposed to solve the above coupling problem, which could be solved by classical fourth-order Runge-Kutta method through MATLAB language programming. Take the organic working fluid R123 for example, the numerical results were verified by experimental results. The numerical results of flow rate, power output, volumetric and isentropic efficiency were in good agreement with the experimental results at the rotation speed of 3000±10rpm under different intake pressure. Then three organic working fluids R123, R245fa and R134a were chose to simulate the characteristics of single screw expanders. Results show that the highest efficiency is R123, followed by R245fa and the last is R134a at the same rotation speed and intake conditions.

1. INTRODUCTION

Due to the energy crisis and environmental issues, much research has been conducted on renewable energy and industrial waste energy recovery technologies during recent years. Among them, organic Rankine cycle (ORC) power generation systems using organic fluids in recovering middle and low-grade energy resources has increasingly attracted attention. The selection of the expander is one of the critical factors which has great influence on cost and efficiency of ORC systems. Volumetric expanders have been proven to be suitable for small size ORC heat recovery systems with the net power output of 1-50 kW (Imran *et al.*, 2016). As a type of volumetric expander, single screw expanders (SSEs) can be applied to low to medium grade heat and waste heat recovery utilizing the ORC system.

However, internal irreversible loss including leakage, heat transfer and friction power loss has great influence on the performances of SSEs. In view of the research on this aspect, some experimental and theoretical research have been carried out. And two methods are commonly used for the theoretical research. One is by establishing the thermodynamics working process models of SSEs, which includes governing equations of control volume, leakage flow models, heat transfer loss and friction loss in the SSEs, Ziviani *et al.* (2014) established a comprehensive simulation model for an SSE, and their model was proved by experimental data obtained from an ORC set-up that employed SES36 as the working fluid. The model agreed with the experimental results within 10% and 15% for the mass flow rate and power output, respectively. Ziviani *et al.* (2017) established a non-symmetric modelling approach to obtain a more comprehensive model of the single-screw machine and the model predicted a maximum mechanical efficiency of approximately 70%. Shen *et al.* (2017) established a mathematical model of different leakage paths and leakage model of the SSE to analyze the internal leakage of SSEs. The other is by establishing semi-empirical models of SSEs. Instead of making clear the internal thermodynamics working process, the semi-empirical models describe the expansion process with the related equations and empirical parameters from experiments. Ziviani *et al.* (2016) studied a small-scale ORC unit for waste heat recovery by employing two working fluids, i.e., SES36 and R245fa and a semi-empirical model has been developed and calibrated to break down the expander internal losses in the case of R245fa. Giuffrida *et al.* (2016) built a semi-empirical modelling of a SSE for small scale ORC to analyze the expander performance based on variations of the operating conditions and pay particular attention to the leakage flow rates, the mechanical losses at the shaft and the ambient heat losses by the proposal of a more physically sound modelling, improving the performance simulation of a single-screw expander for which there exists a wide experimental campaign in literature.

Although some present researches of SSEs have been carried out, rare research can use the thermodynamics working process models of SSEs to directly analyze the effect of internal irreversible loss on the performance of expanders in details, and compare the influence of different organic fluids on the performance of expanders. In this paper, a numerical study of a single screw expander was developed to analyze its internal irreversible loss, which was verified by experimental results taking the organic working fluid R123 for example. Then the performance and characteristics analysis of single screw expanders are conducted for evaluating the effect of three different organic working fluids R123, R245fa and R134a.

2. MATHEMATICAL MODEL

The control volume of the single screw expander is formed by the screw groove, profile surfaces of the gaterotor teeth, and inside wall of the shell is shown in Figure 1. The working process of single screw expander mathematical model is developed based on the mass and energy conservation laws. The influences of leakage loss, heat transfer loss, viscosity friction power loss and suction pressure loss are considered in the mathematical model. The following assumptions are made to establish the model of the working process.

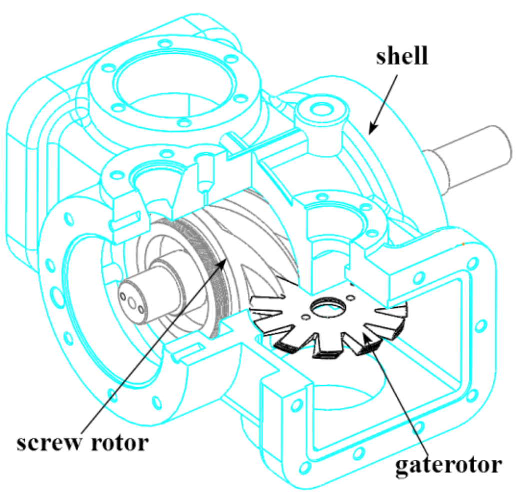

Figure 1: The diagram of single screw expander

(1) 1D flow
(2) Changes in the potential and kinetic energies of the working fluid are neglected
(3) In the control volume, the pressure and temperature of organic gas and lubricating oil are assumed to be homogeneous respectively at any instant
(4) The lubricating oil in the working chamber has no phase change and is treated as an incompressible fluid

2.1 Basic equations for control volume

The mass conservation equation of the working fluid in the control volume during the suction, expansion and discharge process can be expressed as follows:

$$\frac{\mathrm{d}m}{\mathrm{d}\theta_1} = \sum \frac{\mathrm{d}m_{in}}{\mathrm{d}\theta_1} - \sum \frac{\mathrm{d}m_{out}}{\mathrm{d}\theta_1} \tag{1}$$

Where m is the mass flow rate in the screw groove, and m_{in} is the fluid enter and leak in the control volume and m_{out} is the fluid discharge and leaked out of the control volume. The leakage paths in the single screw expander includes nine paths and the structure characteristics of the leakage paths are presented (Shen *et al.*, 2017). A two phase oil-gas mixture flowing through the leakage path of single screw expander, which can be treated as two phase layer flow (Xing *et al.*, 2010), was used in this article. The velocity of gas and oil are calculation from the energy conservation equation:

$$c_{fg} = \sqrt{2\left(h_1 - h_2\right)} \tag{2}$$

$$c_{fl} = c_{fg} / f \tag{3}$$

The leakage mass flow rate of organic fluid is expressed as:

$$\frac{\mathrm{d}m_{lea,i}}{\mathrm{d}\theta} = C_{lea} \alpha A_i \rho_g c_{fg} \tag{4}$$

$$\frac{\mathrm{d}m_{lea,l}}{\mathrm{d}\theta} = C_{lea}(1-\alpha) S_i \rho_l c_{fl} \tag{5}$$

$$f = 0.4 + 0.6\sqrt{\frac{\rho_l}{\rho_g} + 0.4(\frac{1}{\chi} - 1)}\sqrt{1 + 0.4(\frac{1}{\chi} - 1)} \tag{6}$$

$$\alpha = \frac{1}{1 + f\left(\frac{1}{\chi} - 1\right)\frac{\rho_g}{\rho_l}} \tag{7}$$

Where C_{lea} is flow coefficient, which is equal 0.65 in this study (Xing *et al.*, 2010), α is void fraction A_i is the ith leakage path area, ρ_g and ρ_l are the gas and oil density, respectively. c_{fg} and c_{fl} are the gas and oil velocity, respectively. χ is the ratio of gas in the gas-oil mixture. f is slip factor. h_1 and h_2 are specific enthalpy in the high and low pressure chambers, respectively.

Because the suction port area is small and the suction pressure loss cannot be neglected. After considering the suction pressure loss, the suction pressure at the end of suction process is lower than that of ideal suction process, which is treated as stable frictionless flow with no pressure drop. Thus, m_{in} considering the suction pressure loss during the suction process can be calculated by the following equation,

$$\frac{\mathrm{d}m_{in}}{\mathrm{d}\theta_1} = \frac{C_{in} A_{in}}{\omega}\sqrt{2(h_{in} - h_s)} \tag{8}$$

Where A_{in} is the area of suction port, ω is the rotation speed angle of screw rotor, h_{in} and h_s are specific enthalpy obtained from the pressure and temperature at the suction state and the suction chamber, C_{in} is flow coefficient which is equal 0.8 in this article.

The energy conservation equation of the organic gas and lubricating oil in the control volume can be expressed after rearrangement as follows:

$$\frac{\mathrm{d}\left(mu\right)_{CV}}{\mathrm{d}\theta_1} = \sum h_{in}\frac{\mathrm{d}m_{in}}{\mathrm{d}\theta_1} - \sum h_{out}\frac{\mathrm{d}m_{out}}{\mathrm{d}\theta_1} + \frac{\mathrm{d}W}{\mathrm{d}\theta_1} - \frac{\mathrm{d}Q}{\mathrm{d}\theta_1} \tag{9}$$

Where u is the specific internal energy of working fluid, h_{in} and h_{out} are the inlet and outlet specific enthalpies of working fluid, respectively, W is the output power of expander, Q is the heat transfer between working fluid and the boundary of the control volume and lubricating oil film. And the heat can be calculated by

$$\frac{\mathrm{d}Q}{\mathrm{d}\theta_1}=\frac{\kappa A_w\left(T-T_w\right)}{w} \tag{10}$$

Where κ is the heat convection coefficient, A_w is the heat transfer area including bottom, left and right flank of screw groove, and the gaterotor tooth in the control volume. T_w is the temperature of the control volume wall.

By combining thermodynamics properties of the working fluid into equation (1) and (9). The pressure and temperature in the working chamber can be changed into

$$\frac{\mathrm{d}p}{\mathrm{d}\theta_1}=\frac{\frac{1}{v}\frac{\mathrm{d}v}{\mathrm{d}\theta}\left[\left(\frac{\partial h}{\partial v}\right)_T-p-\left(\frac{\partial h}{\partial T}\right)_v\left(\frac{\partial p}{\partial v}\right)_T\left(\frac{\partial T}{\partial p}\right)_v\right]-\frac{1}{V}\left[\sum\frac{\mathrm{d}m_{in}}{\mathrm{d}\theta_1}\left(h_{in}-h\right)+\frac{\mathrm{d}Q}{\mathrm{d}\theta_1}-\frac{\mathrm{d}W_f}{\mathrm{d}\theta_1}\right]-\frac{p}{m}\frac{\mathrm{d}m}{\mathrm{d}\theta_1}}{2-\frac{1}{v}\left(\frac{\partial h}{\partial T}\right)_v\left(\frac{\partial T}{\partial p}\right)_v} \tag{11}$$

$$\frac{\mathrm{d}T}{\mathrm{d}\theta_1}=\frac{\frac{\mathrm{d}v}{\mathrm{d}\theta}\left[\frac{1}{v}\left(\frac{\partial h}{\partial v}\right)_T-2\left(\frac{\partial p}{\partial v}\right)_T-\frac{p}{v}\right]-\frac{1}{V}\left[\sum\frac{\mathrm{d}m_{in}}{\mathrm{d}\theta_1}\left(h_{in}-h\right)+\frac{\mathrm{d}Q}{\mathrm{d}\theta_1}-\frac{\mathrm{d}W_f}{\mathrm{d}\theta_1}\right]-\frac{p}{m}\frac{\mathrm{d}m}{\mathrm{d}\theta_1}}{2\left(\frac{\partial p}{\partial T}\right)_v-\frac{1}{v}\left(\frac{\partial h}{\partial T}\right)_v} \tag{12}$$

$$\frac{\mathrm{d}T_l}{\mathrm{d}\theta_1}=\frac{1}{m_l}\left(T_{lin}\sum\frac{\mathrm{d}m_{lin}}{\mathrm{d}\theta_1}-T_l\sum\frac{\mathrm{d}m_{lout}}{\mathrm{d}\theta_1}-T_l\sum\frac{\mathrm{d}m_l}{\mathrm{d}\theta_1}-\frac{1}{C_l}\frac{\mathrm{d}Q_l}{\mathrm{d}\theta_1}\right) \tag{13}$$

Where u is the specific internal energy of working fluid, h is the specific enthalpies of working fluid, W and W_f is the output power and viscosity power loss of expander, Q, Q_l is the heat transfer between working fluid and the boundary of the control volume and lubricating oil film, respectively. m_{lin}, m_{lout} is the lubricating oil leak in and out of the control volume.

In the simulation, equation (1) is applied to monitor the mass change of the working fluid in the working chambers. equations (11)-(13) is applied to monitor the change of pressure and temperature in the working chamber. According to these functions, the basic state parameters in the working chambers can be obtained after considering the leakage, heat transfer and viscosity loss. Thus, the density, pressure and temperature in the control volume from intake to discharge process also can be obtained, which is most important for solving the power output and efficiency.

The viscosity power loss can be expressed as follows:

$$\frac{\mathrm{d}W_f}{\mathrm{d}\theta_1}=\frac{\sum_{i=1}^{9}\int_0^{b_i}\tau_f c_f L_i \mathrm{d}x}{w} \tag{14}$$

Where μ is the mixture dynamic viscosity, b_i is the i_{th} leakage path width, L_i is the i_{th} leakage path length, c_f is the fluid velocity in the clearance, and τ_f is the viscosity shear force.

2.2 Efficiency

The volumetric efficiency is used to evaluate the expander the leakage effect, which is the rate of geometric structure utilization of the expanders.
The volumetric efficiency can be expressed as follows:

$$\eta_v=\frac{m_{f,th}}{m_{f,real}} \tag{15}$$

Isentropic efficiency η_{iso} is the ratio of the indicated expansion power output W to the isentropic expansion work output W_{iso}. The isentropic efficiency is given by

$$\eta_{iso} = \frac{W}{W_{iso}} \tag{16}$$

The isentropic expansion power output W_{iso} is the ideal power generation of the working fluid through the isentropic process.

$$W_{iso} = m_f \left(h_s - h_{d,iso} \right) \tag{17}$$

Where h_s is the specific enthalpy of the working fluid at the expander inlet, $h_{d,iso}$ is the specific enthalpy of the working fluid at the expander outlet during the isentropic process.
The indicated expansion power output W in the numerical model is calculated by

$$W = -m_{f,real} \int_{p_s}^{p_d} v \mathrm{d}p \tag{18}$$

Where p_s is the pressure of the working fluid at the expander inlet, p_d is the pressure of the working fluid at the expander outlet. The integration is the area circulated by the expansion process in the *p-v* diagram of the single screw expander.

2.3 Calculation method

A separation approach was proposed to solve the above coupling problem, which could be solved by classical fourth-order Runge-Kutta method through MATLAB language programming. Firstly, the expander geometry parameters and operating conditions are input to formulate the mathematic model. Then the governing equations (1)-(14) can be solved considering leakage, suction pressure, heat transfer and viscosity power loss. The pressure and temperature as a function of screw rotor rotation angle and the *p-V* indicator diagram can be obtained. Finally, the expander performance of the working process can be calculated using equations (15)-(18).

3. RESULTS AND DISCCUSION

Take the organic working fluid R123 for example, the numerical results were verified by experimental results (Lei *et al.*, 2016). Figure 2 displays a schematic diagram of the experimental ORC system (Lei *et al.*, 2016), which mainly comprises a single screw expander, evaporator, condenser, working fluid pump, oil separator, liquid container and flowmeters. The liquid R123 in the container is delivered into the evaporator by the working fluid pump, and the liquid R123 absorbs heat in the evaporator and transform the high pressure-saturated or superheated vapor, then flow into the single screw expander to convert pressure into mechanical energy and generate power. After that, the lower pressure R123 sequentially flows into the oil separator and condenser, and become cooled liquid again, which can be recirculated by the working fluid pump. In the R123 circuit, some main parameters can be measured, such as flow rate of R123, torque, pressure and temperature at the inlet or outlet of the expander and so on.

Figure 2: A picture of the ORC system using the single screw expander

Thus, the thermodynamics performance parameters of the single screw expander can be obtained to validate the simulation results. Table 1 shows the numerical results of flow rate, power output, volumetric and isentropic efficiency are in good agreement with the experimental results at the rotation speed of 3000±10rpm under different intake pressure. It is shown that the relative errors between the experimental and simulation results are less than 14%.

Table 1: Comparison between the experimental results and simulation results

Operating parameters			Experiment results				Simulation results				Relative error			
T_s	P_s	P_d	$m_{f,in}$	W	η_s	η_v	$m_{f,in}$	W	η_s	η_v	$m_{f,in}$	W	η_s	η_v
(K)	(bar)	(bar)	(kg/s)	(kW)	-	-	(kg/s)	(kW)	-	-	%	%	%	%
376.55	6.75	1.13	0.235	3.9	0.75	0.81	0.244	4.4	0.85	0.82	4.1	11.9	13.9	1.3
379.15	7.04	1.08	0.244	4.2	0.75	0.80	0.254	4.6	0.83	0.81	4.4	9.3	10.4	1.2
390.85	8.82	1.30	0.305	5.7	0.77	0.80	0.323	6.3	0.83	0.81	5.8	9.5	7.6	1.3
390.15	9.71	1.28	0.345	6.7	0.76	0.81	0.359	7.1	0.83	0.81	4.1	6.1	8.6	0.3
394.35	10.51	1.22	0.372	7.6	0.77	0.81	0.390	8.0	0.81	0.81	5.0	4.7	4.8	-0.1
399.25	11.51	1.41	0.397	8.2	0.81	0.83	0.432	9.1	0.83	0.81	8.7	10.9	2.5	-2.9

A suitable selection of the working fluids is an important factor for the performance and efficiency of an organic Rankine cycle system. In this study, firstly, three different working fluids R123, R245fa and R134a are selected to analyze the characteristics of single screw expanders at the same rotation speed (3000rpm) and intake conditions (intake pressure varies from 0.8MPa-1.3MPa and corresponding saturation temperature with superheat temperature of 5K). Then, the characteristics of single screw expanders at the same intake conditions (0.6MPa, 373K) and different rotation speed (2000rpm-4000rpm) used these three different working fluids are analyzed. Figure 3 and Figure 4 show the specific volume and enthalpy of the three working fluids under the same intake conditions. It can be seen that all of the specific volume of working fluids decrease with the inlet pressure. And the largest specific volume is R134a, followed by R245fa and the last is R123 under the same inlet pressure and temperature, namely the density of 134a is smaller than R245fa and R123 at the same intake conditions. It also can be obtained the specific enthalpy of R245fa and 134a are nearly equal, but that of R123 is relatively lower at the same intake pressure.

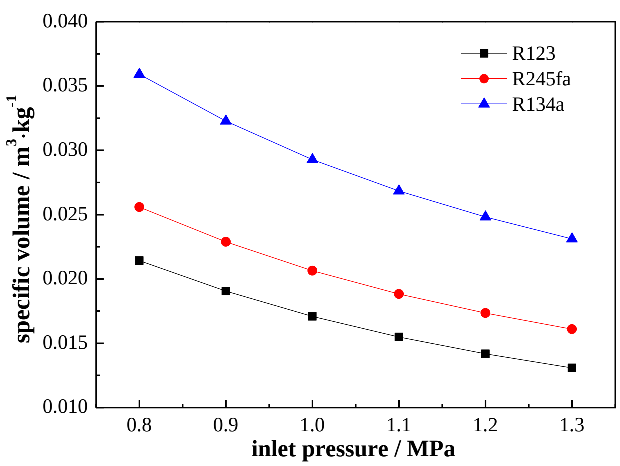

Figure 3: Variation in specific volume with the inlet pressure

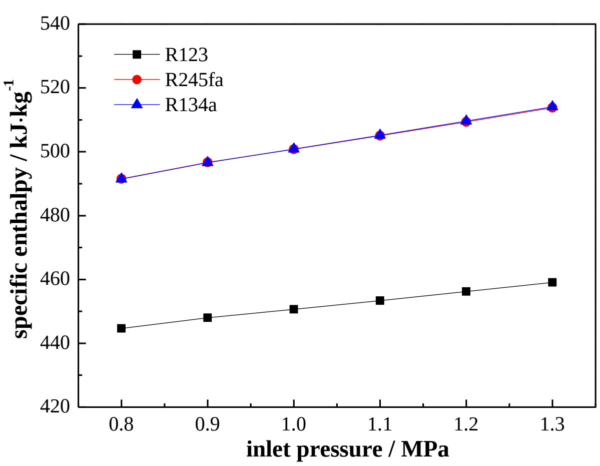

Figure 4: Variation in specific enthalpy with the inlet pressure

The *T*-s diagram of these working fluids are shown in Figure 5. The R134a is wet fluid, while the R123 and R245fa are dry fluid based on the saturated vaporization line in the diagram of *T*-s. The wet fluid leads to droplets at the end of the expansion. Besides, some important factors of the working fluids should be considered, such as the chemical stability, safety, environment-friendly, thermophysical properties, which are listed in Table 2. R123 and R245fa is more toxic than R134a. All of these three working fluids are nonflammable and does not support combustion. The ozone depleting potential (ODP) of R123 and R245fa is null, that of R134a is also close to zero, since the non-null ODP fluids are gradually being terminated by the Montreal Protocol. The greenhouse warming potential (GWP) of R134a is higher than the other two organic fluids, but there is no legislation restricting the use of high GWP organic fluids. For the boil temperature, R134a is as low as -26.3 ℃, which could reduce the cooling water temperature in the condenser and result in a more demanding selection of the condenser. Due to the chemical decomposition and deteriorations of the organic fluids under a high pressure and temperature, a working fluid should be operated under suitable working conditions. It is difficult to define an optimum for each specific thermophysical property independently. Generally, the solutions are obtained by comparing the organic Rankine cycle efficiency or output power of a thermodynamic model.

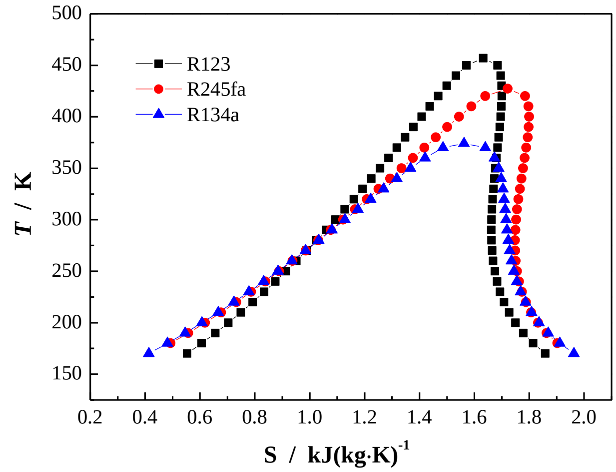

Figure 5: *T*-s diagram of the working fluids

Table 2: The basic parameters of organic fluids

	R123	R245fa	R134a
Safety	B1	B1	A1
ODP	0.02	0	0
GWP	93	820	1300
T-boil (℃)	27.85	15.3	-26.2
P-critical (MPa)	3.66	3.65	4.07
T-critical (℃)	183.68	154.01	101.1

Figure 6 shows mass flow rate of different working fluid changes with the inlet pressure. All of the mass flow rate of working fluids increase linearly with the inlet pressure. This is because the density of the working fluid can be enhanced by enlarging the inlet pressure. However, the largest mass flow rate is R123, followed by R245fa and the last is R134a under the same inlet pressure, which is one of the main reason leading to the volumetric efficiency has the similar relationship at the same inlet pressure, as shown in Figure 7. The volumetric efficiency of different working fluid changes with the inlet pressure. It is observed that the volumetric efficiency remains almost unchanged with increase in inlet pressure, the volumetric efficiency range of R123, R245fa and R134a is 80.78-81.73%, 79.73-80.63% and 77.20-78.04%, respectively. This is because the working fluid density increase with the increase in the inlet pressure. Thus, the theoretical mass flow rate increases with inlet pressure. Although the higher pressure difference allows more leakage under higher inlet pressure, theoretical and real mass flow rate also increase, so the volumetric efficiency has hardly changed.

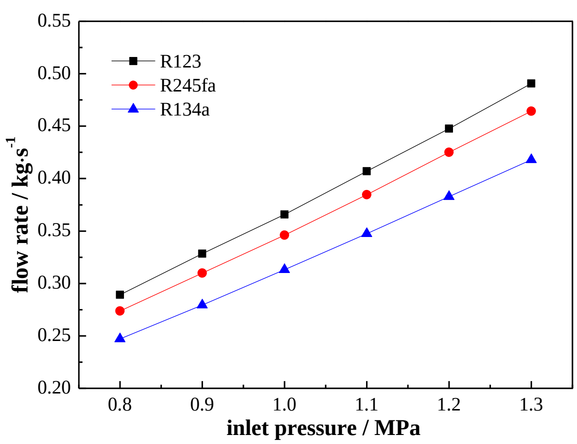

Figure 6: Variation in flow rate with the inlet pressure

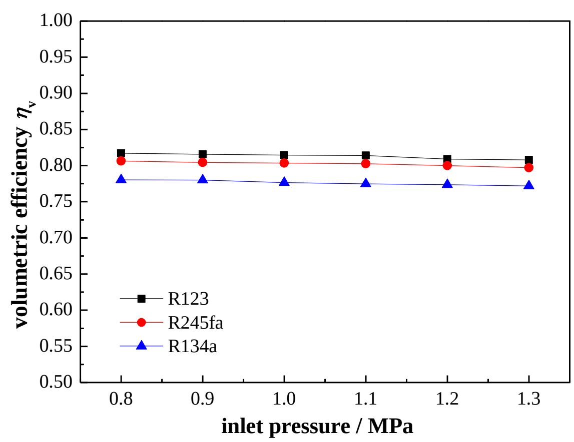

Figure 7: Variation in volumetric efficiency with the inlet pressure

The variation in power output with the inlet pressure by simulation is shown in Figure 8. The power output of each working fluids increases with the increase in the inlet pressure. And power output increased 6.1-10.5kW, when the inlet pressure is from 0.8 MPa to 1.3 MPa. The reason is that with the increase in the inlet pressure, the density and inlet flow rate increase, which means working fluid output more power in the control volume. But when the inlet pressure is identical, the power output of these three working fluids are almost same, especially for R245fa and R134a, that is because the specific enthalpy of R245fa and 134a are nearly equal as shown in Figure 4. Figure 9 shows isentropic efficiency of different working fluid varies with the inlet pressure. The largest isentropic efficiency of R123, R245fa and R134a is at the inlet pressure of 0.8MPa and the value is 78.41%, 69.67% and 54.50%, respectively. The isentropic efficiency of each working fluid decreases slightly with the increase of the inlet pressure. Although the power output is almost same at the same inlet conditions, the isentropic efficiency differ greatly and the relation from large to small is R123, R245fa, R134a.

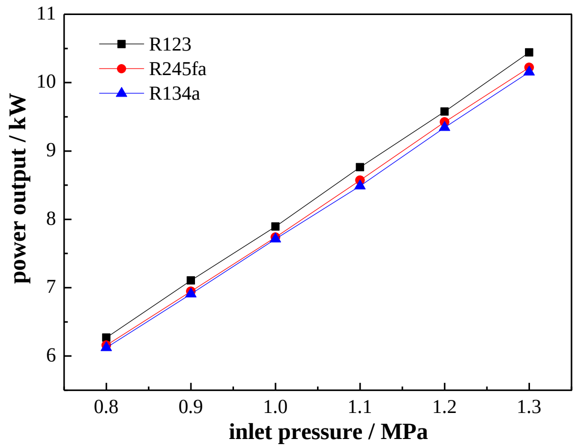

Figure 8: Variation in power output with the inlet pressure

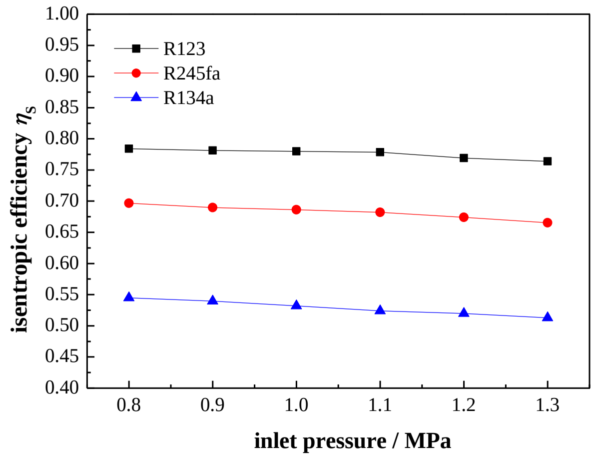

Figure 9: Variation in isentropic efficiency with the inlet pressure

The following is given the performance of single screw expanders at the same intake conditions (0.6MPa, 373K) and different rotation speed (2000rpm-4000rpm) used these three different working fluids R123, R245fa and R134a. The volumetric efficiency of different working fluid changes with the rotation speed, as shown in Figure 10. It is observed that the volumetric efficiency increases with increase in rotation speed, the volumetric efficiency range of R123, R245fa and R134a is 74.61-91.81%, 74.93-90.4% and 69.38-85.38%, respectively. This is because the working fluid flow rate increases with the increase in the rotation speed. Moreover, the higher rotation speed allows relatively less leakage, thus, its volumetric efficiency is relatively high.

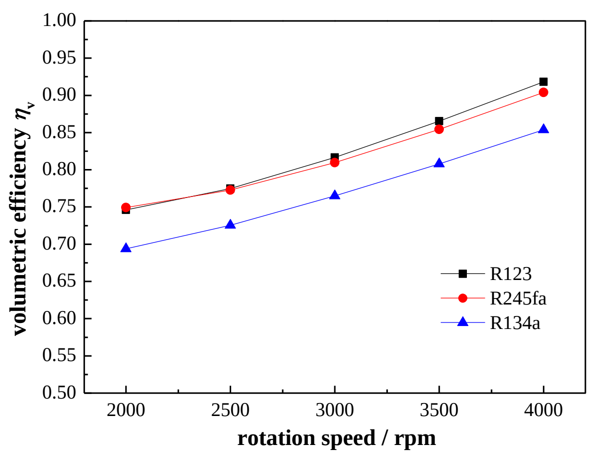

Figure 10: Variation in volumetric efficiency with the rotation speed

Figure 11 shows power output of different working fluid varies with the rotation speed. The power output of each working fluids increases linearly with the increase in the rotation speed. And power output increased 2.7-6.7kW, when the rotation speed is from 2000rpm to 4000rpm. Because the working fluid flow rate increases with the rotation speed, which means working fluid output more power in the control volume. As discussed in Figure 8, when the rotation speed is same, the power output of R245fa and R134a are almost same, this is because the specific enthalpy of R245fa and 134a are nearly equal, which is shown in Figure 4. The variation in isentropic efficiency of different working fluid with the rotation speed is shown in Figure 12. Due to the increase of the power output and the flow rate with the increase in the rotation speed, the isentropic efficiency of each working fluid increases. But the isentropic efficiency growth become slow down when the rotation speed raise, the main reason is that mechanical friction increase sharply with the rotation speed. It also can be observed that the isentropic efficiency under same rotation speed from large to small is R123, R245fa, R134a. This is because that the density of R123 is largest, followed by R245fa, and 134a is smallest at the same intake conditions, which also can be seen in Figure 3.

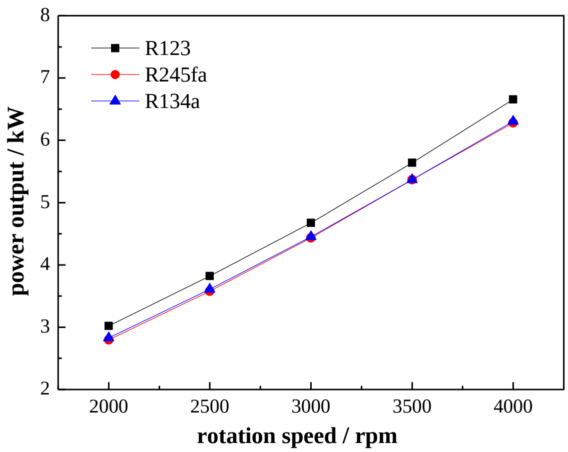

Figure 11: Variation in power output with the rotation speed

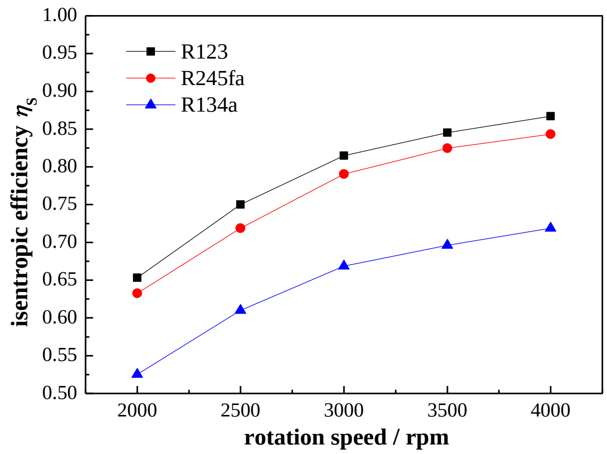

Figure 12: Variation in isentropic efficiency with the rotation speed

4. CONCLUSIONS

The presented simulation results of the single screw expander are given in the paper. The calculated results are overall in good agreements with the experimental results. Three different working fluids R123, R245fa and R134a are selected to analyze the characteristics of single screw expanders. It accounts for effects like suction pressure, heat transfer, leakage, viscous friction power loss and the performance characteristics of the single screw expander are examined. For the given SSE, the highest efficiency is R123, followed by R245fa and the last is R134a at the same rotation speed and inlet conditions. Besides, it can be seen from the cases under investigation, expander efficiency can be optimized by selecting a proper working organic fluid operated at suitable working conditions, which also has great influence on the organic Rankine cycle system efficiency.

NOMENCLATURE

Variables

A	area (m)	b	width of the leakage path (m)
C	flow coefficient	c_f	speed of fluid (m s^{-1})
c_l	specific heat of oil (J kg^{-1} K^{-1})	d_g	equivalent diameter (m)
f	slip factor	h	specific enthalpy(J kg^{-1})
i	gear ratio	L	leakage path length (m)
m	mass (kg)	m_f	mass flow rate (kg s^{-1})
n	rotational speed (rpm)	p	pressure (Pa)
Q	heat transfer quantity(J)	T	temperature (K)
U	internal energy(J)	u	specific internal energy(J kg^{-1})

V	volume (m^3)	v	specific volume(m^3 kg^{-1})
W	work (J)	w	rotation angle speed (rad s^{-1})

Greeks

α	void fraction	θ	rotation angle (rad)
λ	heat conductivity	μ	dynamic viscosity (Pa s)
η	efficiency	ρ	working fluid density (kg m^{-3})
κ	heat convection coefficient	τ_f	viscous shear force
χ	ratio of gas in the gas-oil mixture		

Subscripts

1	screw	*CV*	control volume
d	discharge	*g*	gas
i	the i_{th} leakage path	*iso*	isentropic
lea	leakage	*in*	fluid enter the control volume
l	lubricating oil	*out*	fluid leave the control volume
real	actual value	*s*	suction
th	theoretical value	*v*	volumetric

REFERENCES

Giuffrida, A. (2017). Improving the semi-empirical modelling of a single-screw expander for small organic Rankine cycles. *Appl. Energy*, 193, 356-368.

Lei, B., Wang, W., Wu, Y. T., et al (2016). Development and experimental study on a single screw expander integrated into an Organic Rankine Cycle. *Energy*, 116, 43-52.

Shen, L. L., Wang, W., Wu, Y. T., et al (2017). Theoretical and experimental analyses of the internal leakage in single-screw expanders. *Int. J. Refrig.*, 86, 273-281.

Imran, M., Usman, M., Park, B. S., et al (2016). Volumetric expanders for low grade heat and waste heat recovery applications. *Renew. Sus. Energy Rev.*, 57, 1090-109.

Xing, Z. W., Peng, X. Y., Shu, P. C. (2010). Study of Leakage Characteristics in Twin Screw Compressor with Refrigerant R134a. *Int. J. Refrig.*, 4, 23-28.

Ziviani, D., Bell, I. H., Broek, M. V. D., et al (2014). Comprehensive model of a single screw expander for ORC-systems. *Proceedings of the 22nd International Compressor Engineering Conference at Purdue* (pages1-10), Purdue

Ziviani, D., Gusev, S., Lecompte, S., et al (2017). Optimizing the performance of small-scale organic Rankine cycle that utilizes a single-screw expander. *Appl. Energy*, 189, 416-32.

Ziviani, D., Gusev, S., Lecompte, S., et al (2016). Characterizing the performance of a single-screw expander in a small-scale organic Rankine cycle for waste heat recovery. *Appl. Energy*, 181, 155-70.

ACKNOWLEDGEMENT

This work was supported by the National Key R&D Program of China (grant number 2016YFE0124900).